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CORRELATION OF EVAPORATIVE HEAT TRANSFER COEFFICIENTS FOR REFRIGERANT MIXTURES

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ABSTRACT

On the basis of the measured evaporative heat transfer coefficients for pure refrigerants (R22, R32, R125, R134a, R290, and R600a) and refrigerant mixtures (R32/R134a and R290/R600a) in a horizontal tube, a correlation of evaporative heat transfer coefficients for refrigerant mixtures is developed. An analysis of evaporative heat transfer is performed for an annular flow of refrigerant mixtures. Mass transfer effect due to composition difference between liquid and vapor phases, which is considered as a driving force for mass transfer at the interface, is included in this analysis. Correction factor C_F is introduced to the correlation for refrigerant mixtures. The evaporative heat transfer coefficients are calculated using the correlation considering nucleate boiling effect in the low quality region and mass transfer effect for zeotropic refrigerant mixtures.

INTRODUCTION

New alternative refrigerants are in production and the application techniques are being developed. As higher efficiency of the system is constantly required, a new technique for performance enhancement is pursued and the use of refrigerant mixtures is thought to be a solution. Some refrigerant mixtures show temperature glide during evaporation and condensation (i.e., zeotrope), and this temperature glide can be utilized to reduce the mean temperature difference between the secondary heat transfer fluid and refrigerant, which is known to increase the cycle efficiency from the basic principles of thermodynamics.

It has been known by many researchers that the heat transfer coefficient (HTC) of zeotropic refrigerant mixtures is smaller than linearly interpolated HTC based on those of pure constitutive refrigerants. This phenomenon is attributed to non-linear behavior of thermodynamic properties of refrigerant mixtures and mass transfer effect caused by composition change during the evaporation of zeotropic refrigerant mixtures.⁽¹⁻⁴⁾ For mixtures, mass transfer resistance is introduced to the correlation that is obtained by regression of experimental data. However, since the coefficients in the correlation are determined based on the experimental data, the use of such correlations becomes limited when they are applied to other fluids. Therefore, simple and physically meaningful correlations that can be used to many fluids are required. Furthermore, it is desired that the correlations can be applied for mixtures.

In this study, a correlation which can predict the HTCs of zeotropic refrigerant mixtures will be presented. To consider the different characteristics between pure refrigerants and refrigerant mixtures, a factor for the mass transfer resistance caused by composition difference of liquid and vapor phases during the evaporation will be introduced. Correlations and experimental data of evaporative heat transfer will be compared.

EXPERIMENTS

Test Review and Data Reduction

The HTCs for pure refrigerants and refrigerant mixtures flowing in a horizontal tube have been

measured under uniform heat flux condition by applying electric current directly to the tube. Heat transfer coefficients at different mass fluxes and heat fluxes were measured and the dependence on the overall compositions for mixtures has been investigated.⁽⁵⁾

In order to calculate the HTC's, local heat flux and the temperatures of the refrigerant and the wall should be measured. In our previous study, outer wall temperature was measured and inner wall temperature was estimated considering radial conduction. Thermocouple tips were fastened on the outer wall by aluminum tape at 4 positions around the tube and the lead wire was attached on the wall along the axial direction to avoid axial conduction through the wire from the surroundings. The refrigerant temperature was determined by measuring the saturation pressure.

Heat transferred to the refrigerant at the test section was measured by watt meter. The output signals of the thermocouples and pressure transducers were carried to the personal computer through multi-channel recorder using the GPIB interface interface. Estimated error was about 0.2 °C for the outer wall temperature measurement and 0.5 °C for the measurement of saturation temperature of the refrigerant.

Heat transfer coefficient is defined as in Equation (1).

$$h_k = \frac{q''}{T_{wi,k} - T_s}, \quad (1)$$

where q'' is heat flux and T_{wi} , T_s represent inner wall and saturation temperature, respectively. Subscript k represents top, bottom, left, and right sides, respectively. For mixtures, thermodynamic quality which can be determined by energy balance should be known to calculate saturation temperature. Local heat transfer coefficient, h , at local positions is determined by averaging above mentioned 4 temperatures. Experimental data for deducing the correlation of this study is listed in reference (5). The error in measuring heat transfer coefficients is about 5.8% in our previous study.

Comparison of Existing Correlations for Pure Substances with Experimental Data

Root mean square (RMS) values of relative errors between existing correlations and experimental data with respect to quality for pure refrigerants such as R22, R32, R125, R134a, R290, and R600a are presented in Table 1. The numbers in parentheses represent the result with data for quality range over 0.2. The fact that the error for the quality range over 0.2 becomes smaller implies that existing correlations can not represent well the influence of nucleate boiling in the low quality region.

Among the existing correlations, the convective boiling term in the Chen's correlation is in good agreement with the experimental data except for the early period of evaporation (for the quality range over 0.2). This implies that the convective boiling could be represented quite well with Chen's convective boiling term. This will be considered in developing new correlation for refrigerant mixtures.

Table 1 Root mean square error (%) between predicted and experimental heat transfer coefficients (numbers in parenthesis represent RMS error for the experimental data of quality over 0.2)

Refrigerants	Chen ⁽⁶⁾	Shah ⁽⁷⁾	Gungor & Winterton ⁽⁸⁾	Kandlikar ⁽⁹⁾
R22	32.0 (10.7)	17.2 (13.9)	29.9 (15.6)	22.8 (18.9)
R32	43.2 (8.6)	13.4 (10.8)	39.4 (13.2)	29.5 (25.2)
R134a	77.8 (13.6)	19.4 (15.7)	72.8 (17.2)	23.8 (20.9)
R125	35.0 (12.5)	12.4 (10.9)	31.2 (14.2)	18.0 (18.2)
R290	34.8 (16.4)	19.3 (13.8)	32.3 (14.1)	30.8 (30.8)
R600a	34.8 (15.6)	35.2 (24.4)	38.1 (28.0)	38.5 (38.5)

ANNULAR FLOW ANALYSIS

In modeling the evaporative heat transfer in an annular flow regime, constant heat flux condition is assumed. The influence of nucleate boiling in the low quality region is neglected and the thickness of liquid layer is assumed to be constant in the circumferential direction. The entrainment to vapor phase or the deposition to liquid phase are not considered in this analysis. In Fig. 1, X and Y represent liquid and vapor compositions of more volatile component and T represents temperature, J , mass flux, and Q , heat transfer rate, respectively. Subscripts l , v , i , b , and wi imply liquid phase, vapor phase, interface, bulk, and inner wall, respectively. Mass conservation for more volatile component at the interface is represented as in Equation (2) and shown in Fig. 1.

$$-m_i''X_i - J_l = -m_i'Y_i + J_v, \quad (2)$$

where m_i'' represents evaporated mass flux. For zeotropic refrigerant mixtures, the vapor phase temperature in two phase flow is calculated by using equation of state. Interface temperature is determined so as to satisfy the condition that the mass flux in Equation (2) is the same as the mass flux calculated from the overall energy and mass balance equations. Once interface temperature is determined, the wall temperature is automatically determined from the heat transfer relation in the liquid layer.

Based on the above analysis, a new modification coefficient C_F as shown in Equation (3) was defined to consider the degradation of HTC's due to the temperature difference between interface and vapor phase during the evaporative heat transfer process of zeotropic refrigerant mixtures.

$$\frac{\Delta T}{\Delta T + \Delta T_i} = 1 - C_F, \quad (3)$$

where $\Delta T = T_w - T_i$, and $\Delta T_i = T_i - T_v$.

DEVELOPMENT OF HEAT TRANSFER CORRELATION FOR REFRIGERANT MIXTURES

Evaporative Heat Transfer Correlations for Pure Refrigerants

In this study, the Chen correlation shown in Equation (4) is chosen to be the basic model for evaporative heat transfer correlation.

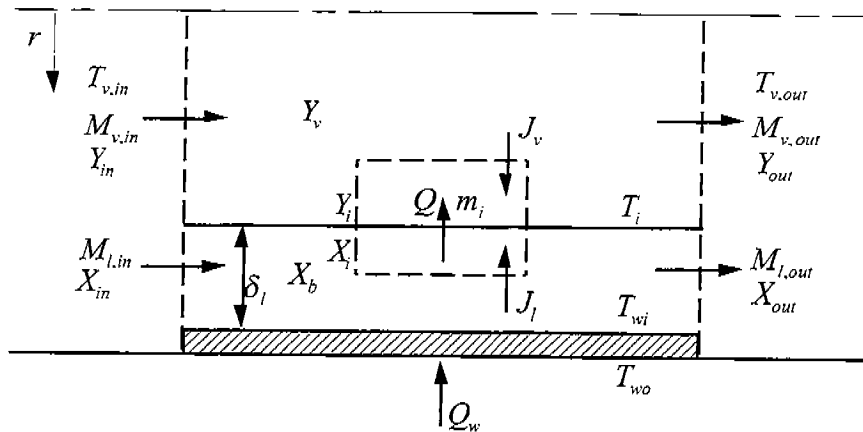


Fig. 1 Control volume for the annular flow analysis in this study.

$$h_{TP} = S \cdot h_{NB} + F \cdot h_{conv} . \quad (4)$$

Measured heat transfer coefficient, h_{meas} can be expressed as a combination of h_{NB} , the influence of nucleate boiling and h_{CB} , that of convective boiling, as shown in Equation (5).

$$\begin{aligned} h_{meas} &= h_{NB} + h_{CB} \\ &= S \cdot h_{pool} + F \cdot h_{conv} , \end{aligned} \quad (5)$$

where S and F represent the suppression and enhancement factors. Once convective boiling term is determined, nucleate boiling term can be calculated by subtracting h_{CB} from h_{meas} . Important factors in nucleate boiling are the flow condition and heat flux condition. Therefore, S in Equation (5) is set as a function of Martinelli parameter (X_{tt}) and non-dimensional boiling number (B_o) as shown in Equation (6). The correlation of Cooper⁽¹⁰⁾ is used to obtain h_{pool} in the nucleate boiling region. The coefficients a_0 , a_1 , and a_2 are determined by least square fitting as 0.0592, 0.668, and 1.38, respectively.

$$S = a_0 X_{tt}^{a_1} B_o^{a_2} . \quad (6)$$

Fig. 2 shows the deviations of heat transfer coefficients calculated from modified correlation equation with experiment data. Modified correlation equation is expressed as in Equation (5) with S in Equation (6). It predicts experiment data within 18.9% error bound for every condition of heat flux, mass flux, and operation pressure in this study. Satisfied results are obtained for pure refrigerants with modified correlation equation. Since complicated correlation equation is difficult to use and the coefficients are sometimes hard to determine, the correlation equation of simple form is chosen in this study and this modified equation represents the experimental data pretty well. The main reason to introduce the correlation equation for pure refrigerants is that the same form of equation will be used for mixtures by modifying the factor F according to the annular flow analysis.

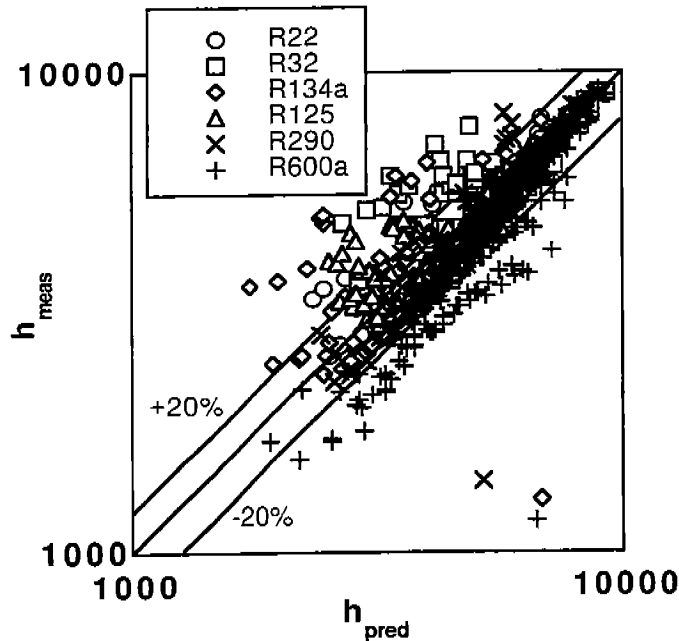


Fig. 2 Comparison between predicted and experimental heat transfer coefficients for pure refrigerants

Correlation of Evaporative Heat Transfer Coefficient for Zeotropic Refrigerant Mixtures

C_F in Equation (3) has been curve-fitted based on the annular flow analysis in the form of Equation (7) by introducing $|Y - X|$, which is the composition difference between vapor and liquid phases. Curve-fitted coefficients, A , n for zeotropic refrigerant mixture, R32/134a are 0.569, 0.860; for R290/600a, A , n are 0.533, 0.828, respectively

$$C_F = A|Y - X|^n \quad (7)$$

HTCs of refrigerant mixtures can be calculated considering the mixture effect with the F factor in Chen's correlation as in Equation (8).

$$h = F_m \cdot h_{conv} \quad (8a)$$

$$F_m = (1 - C_F) \cdot F \quad (8b)$$

F is the Chen's F factor and F_m is a modified factor to predict the HTCs of zeotropic refrigerant mixtures considering the heat transfer degradation due to mass transfer. h_{conv} is the heat transfer coefficient of Dittus-Boelter type. The HTCs of zeotropic refrigerant mixtures can be calculated by defining F_m as in Equation (8b).

When mass transfer effect due to composition difference is introduced and the influence of nucleate boiling is considered, the RMS of relative errors are 8.9% for R32/134a and 14.6% for R290/600a. Fig. 3 shows the comparison of experimental data with heat transfer coefficients calculated from the correlation equation for mixtures in this study. The experimental data for R32/R134a and R290/R600a are obtained for the reduced pressure of 0.15 and mass flux of 424 kg/m²s. The correlative equation for refrigerant mixtures includes the heat transfer enhancement effect caused by nucleate boiling in the low quality region and the heat transfer degradation effect due to mass transfer due to composition change between liquid and vapor phases.

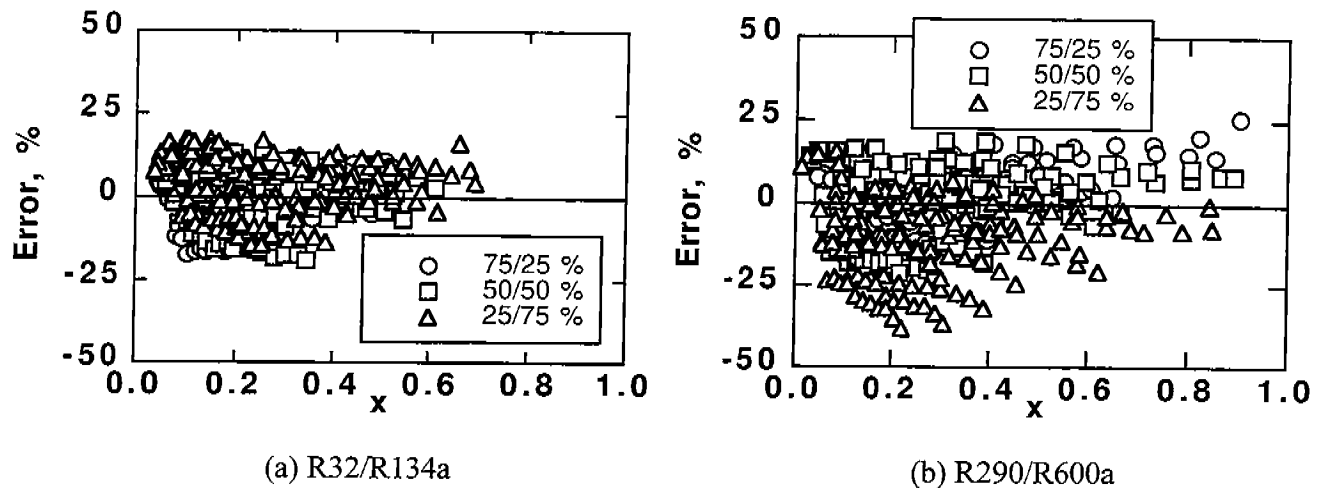


Fig. 3 Relative error between predicted and experimental heat transfer coefficients for refrigerant mixtures

CONCLUDING REMARKS

In order to predict the evaporative heat transfer coefficients for zeotropic refrigerant mixtures in a horizontal tube, annular flow analysis has been done which is the most dominant flow in the evaporator of refrigeration and air-conditioning system. Influence of mass transfer during the evaporative heat transfer process for zeotropic refrigerant mixtures was considered in the analysis. Based on experimental results with several refrigerants, a correlation is developed to predict evaporative heat transfer coefficients for refrigerant mixtures. The correlation equation predicts experiment data for pure refrigerants within 18.9% error bound for all conditions of heat flux, mass flux, and operation pressure in this study.

Based on Chen's correlation, the modification factor, C_p , is introduced by considering the composition difference between liquid and vapor phases for zeotropic refrigerant mixtures. Predicted results show that RMS values of relative errors are 8.9% for R32/134a and 14.6% for R290/600a. The correlation equation in this study is simple to use and predicts the HTC's of refrigerant mixtures quite well. This equation includes the heat transfer enhancement effect caused by nucleate boiling in the low quality region and the heat transfer degradation effect due to the mass transfer resistance because of the composition difference between liquid and vapor phases.

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